HUGHES Research Laboratories

A DIVISION OF HUGHES AIRCRAFT COMPANY
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MALIBU, CALIFORNIA

In reply refer to:
65M-4776/A3268

22 July 1965

TO: Director, Lunar and Planetary Programs
Code SL, Headquarters
National Aeronautics and Space Administration
Washington, D. C.

SUBJECT: Research Contract Status Report Number 9
Contract NASW-1035

In accordance with contract requirements, Hughes Research Laboratories transmits herewith two (2) copies of Contract Status Report Number 9 along with a reproducible copy. The remainder, which is comprised of eighteen (18) reports, is being forwarded under separate cover to the NASA Representative in Bethesda, Maryland.

Based on statements contained in Part II of subject report, the Contractor recommends that it be allowed to proceed with Phase 2 of the contract (construction of the research model). Approval by NASA Headquarters prior to 16 August will enable the Contractor to maintain continuity of contractual effort, thereby alleviating the later need for a contractual time extension.

Further inquiries regarding this report may be directed to Mr. Thomas F. Oliver, Hughes Research Laboratories, Malibu, California, area code 213, telephone GLobe 6-6411, extension 363, TWX No. 213-654-5259.

HUGHES RESEARCH LABORATORIES
DIVISION OF HUGHES AIRCRAFT COMPANY

Thomas F. Oliver

Thomas F. Oliver
Contract Administrator

TFO:ic

Enclosures - 2

Copies to: National Aeronautics and Space Administration
Post Office Box 5700
Bethesda, Maryland 20014
ATTN: NASA Representative (with 18 enclosures)

Contracting Officer, Code BCB
Headquarters
National Aeronautics and Space Administration
Washington, D. C. 20546 (w/o enclosure)

HUGHES Research Laboratories
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22 July 1965

RESEARCH CONTRACT STATUS REPORT

Title: Research on Gravitational Mass Sensors

Contract: NAS W-1035

Period: 15 June 1965 to 15 July 1965

I. ACCOMPLISHMENTS

The vacuum leaks were eliminated from the chamber holding the solidly mounted sensor and the unit was shipped to Cambridge Thermionic Corporation (CAMBION) in Cambridge, Massachusetts, to test its compatibility with their three-axis magnetic support. A visit will be made by Robert L. Forward to CAMBION on 2 August to discuss the results of their tests and to investigate the feasibility of future cooperative effort.

Air leakage through the slip rings into the vacuum chamber has been a continuing problem. It has been possible to stop the leaks by applying generous amounts of Glyptol to the slip rings until they stop leaking and then cleaning the excess off the contacts to restore electrical continuity. However, this is not a satisfactory long-term solution, so some small 8 pin hermetic glass-to-metal feed-throughs were purchased and the slip ring mount is being redesigned so that the vacuum is maintained by the hermetic seal rather than the slip rings.

A new adjustable sensor head assembly has been fabricated and is now being prepared for initial tests. The purpose of the new design is to allow for the separate adjustment of the various sensor parameters such as arm length, frequency and Q in order to achieve better sensor symmetry than is obtainable with the present monolithic designs. The new sensor head consists of a central hub, four arms and four adjustable mass assemblies. The hub is designed to clamp the arms rigidly for good cross coupling and yet allow the arm and mass assembly to be moved in and out about 0.070 inch for mass balance of the final assembly. The arms have a 0.125 inch thick base where they fasten to the hub and an outer bending portion that is 0.030 inch thick and about 0.70 inch long. The adjustable masses, which clamp onto the ends of the arms, consist of two small masses and a double eccentric which can be adjusted to vary the effective length of the arm and yet maintain a center of mass coincidence with the center line of the arm. By moving the masses, the effective length of the arm can be varied from 0.60 to 0.78 inch to yield a frequency shift of several cycles about the design frequency of 100 cps.

The arms are presently being matched in frequency, Q and transducer output separately and will then be installed in the hub for final balancing. Since this sensor structure has a lower frequency of operation and a better geometry factor (see page 36 in Quarterly Progress Report No. 2) than the monolithic designs, the voltage output of this sensor due to the gravitational force gradient of large nearby objects should be in the range of 20 μV.

The single-axis magnetic bearing support and drive fabricated by the University of Virginia is performing satisfactorily. The servo loop has two resonant frequencies, one at 2 cps and the other at 8 cps. These are low enough so that the magnetic support gives excellent vibration isolation at the sensor response frequency of 170 cps. When the servo loop is properly adjusted, the vertical stability of the support is good, except for a long term drift which requires that the servo gain be adjusted every few minutes. No permanent solution has been found to this minor problem. Lateral stability gave some problem for a while as the rotor tended to be stable in a circular track around the center, rather than just at the center. This was eliminated by the insertion of a sponge rubber ring into the damping fluid surrounding the hanging central iron core of the support magnet. This small additional restoring force constrained the iron core to stay at its central position without preventing the motion necessary for lateral damping. An external vacuum chamber is being designed and will be fabricated to eliminate the drag and vibration caused by the rotation of the sensor chamber through the ambient air.

Further noise tests continued on the air bearing and the magnetic bearing using a sensor chamber containing one of the older sensors on a torsion wire mount. The voltages generated by a transducer on one of the arms was taken directly out through the slip rings to a General Radio preamplifier tuned to the sensor frequency (170 cps). Since the primary concern was to find and investigate the sources of noise, there was no attempt made at this stage of the investigation to balance either the sensor or the sensor chamber or to use phase cancellation on the signals from different arms. As was reported in the last monthly report, the results of the static bearing noise tests were that the noise output of a well isolated sensor was about 0.1 μV. When the sensor was sitting on the workbench the noise was about 0.2 μV. With the sensor levitated on the air bearing the noise was about 3.0 μV, and with the sensor levitated on the magnetic bearing, the noise was only 0.15 μV.

Bearing noise tests were then made under dynamic conditions. These tests were limited to rotation speeds of about 3000 rpm in the magnetic bearing support because of air friction. The first tests were made by bringing the sensor up to speed, turning the drive off, and measuring the noise output at 170 cps as the sensor coasted down. The noise output of the rotating sensor on the magnetic bearing started at about 180 μV at 3000 rpm where there is a minor peak and decreased slowly in amplitude to 4 μV at about 300 rpm, with a peak at 1290 rpm (1/4 normal rotation speed) of 180 μV. The noise output from the air bearing, under the same conditions, started at 1200 μV at 3000 rpm and decreased slowly to 24 μV at 300 rpm. The air bearing had a noise peak of 1200 μV, at 1290 rpm as well as other peaks that were not seen on the magnetic bearing. In general, under dynamic as well as static tests, the air bearing is 14 db or more noisy than the magnetic bearing. This excess noise seems to be due to the rush of air through the bearing and there is no obvious way to eliminate the problem. The noise seen on both bearings seems to be primarily due to windage (the interaction of the rotating sensor chamber with the ambient air). This will be eliminated in the magnetic bearing by the addition of the external vacuum chamber, but this is obviously not possible for the air bearing. These tests indicate that the magnetic support is, in general, superior to the air bearing support for our purposes.

Another potential source of vibrational noise, relative motion between the slip rings and brushes, will be eliminated by the design and fabrication of a simple telemetry readout for future designs.

Additional noise tests were made with the magnetic support drive on and two additional sources of noise due to the drive fields were found. One is a general noise level increase due to hash and hum in the drive amplifiers. This causes about 3 μV of noise and can only be seen when the drive amplifiers are turned on, but the sensor is not yet rotating. This noise will be eliminated by filtering the output of the drive amplifiers. The second is a torqueing noise due to the interaction of the rotating drive field and the remnant magnetic poles in the hysteresis plate. This noise is a direct function of the drive power and for synchronous operation at 1290 cps with large drive levels, the noise output at 172 cps can be many millivolts. However, if the drive is lowered to a level just sufficient to maintain synchronous rotation, the noise level drops to the 180 μV level seen under free rotation conditions. This synchronous drive noise can be our most troublesome noise source since the sensor operation requires rotation at half the sensor vibration frequency, but there is good reason to believe that it can be overcome.

Since the drive power requirements are dictated by air friction and it is intended to enclose the sensor vacuum chamber in an external vacuum chamber in the future, the drive power necessary for synchronous operation should be substantially less and the noise should be correspondingly reduced. We are also purchasing an eight-pole motor to see if this will aid in reducing the vibrational noise generation. Another possibility is to drive the motor asynchronously and to control the speed by a feedback loop.

No major problems have arisen which will materially impede the performance of the contract.

II. CONCLUSIONS AND RECOMMENDATIONS

The results of the theoretical and experimental work carried out in the first phase of our program to investigate the feasibility of gravitational mass sensors are:

1. The most promising form of gravitational mass sensor is a cruciform shaped spring-mass system. Experimental results show that its response characteristics are not changed appreciably under the necessary rotation speeds. Theoretical studies indicate that although the sensitivity is slightly lower than that of an ideal sensor because of the use of a strain transducer for sensing the vibratory motion, the voltage outputs calculated for gravitational gradients that can be produced in the laboratory are about 20 μV and should be easily measured with ordinary laboratory equipment.

2. The most promising form of suspension to support the gravitational mass sensor in the earth's gravitational field is a magnetic support. The basic noise (0.15 μV) of the bearing itself is less than one percent of the expected signal levels of 20 μV. The noise introduced by air turbulence and the drive motor in the present test setup during rotation is fairly high, ranging up to 100 times the expected signal level, but the sources of the noise are understood and methods to decrease the effects of these noise sources are known. The newer, more symmetric sensor should be less responsive to the vibrations of the sensor chamber, and the external vacuum chamber should greatly decrease the sensor chamber vibrations caused by air turbulence and motor drive forces. There is every reason to expect, in the remainder of the contract, the effects of noise on the sensor can be reduced to the point where gravitational gradient signals can be seen.

It is recommended that Phase 2 effort proceed with the further study of sensor design, sensor mounting, isolation and readout and drive noise reduction using the present single-axis magnetic bearing and drive. It is further recommended that a three-axis magnetic bearing and drive be purchased with lower characteristic servo frequencies, tighter tolerances on drift and the ability to be oriented in any direction. This magnetic bearing and drive is meant to be the centrifuge and associated equipment provided in the special test equipment section of the Statement of Work. The expected cost of this equipment is $15,000.

III. FUTURE PLANS

The Third Quarterly Progress Report will be written. The new sensor design will be tested in the magnetic bearing under the same static and dynamic conditions as the old sensor to investigate the effects of increased sensor symmetry on sensor susceptibility to vibrational noise. An external vacuum chamber will be fabricated for the magnetic bearing to investigate the effect of the ambient air on noise generation and drive power reduction. The drive amplifiers will be filtered to reduce their contribution to the noise. A simple telemetry readout will be constructed to eliminate slip ring vibrations. After approval to continue with the Phase 2 effort is received, arrangements will be made with either the University of Virginia or CAMBION for the fabrication of a three-axis magnetic support and drive and further sensor head and sensor chamber designs will be constructed.

Prepared by: Robert L. Forward

Robert L. Forward
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